# A new LPV/Hinf semi-active suspension control strategy with performance adaptation to roll behavior based on non linear algebraic road profile estimation

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# A new LPV/$\mathcal{H}_\infty$ semi-active suspension control strategy with performance adaptation to roll behavior based on non linear algebraic road profile estimation

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***Abstract*— This paper presents a new LPV/$\mathcal{H}_\infty$ semi-active suspension control strategy for a commercial vehicle equipped with 4 Magneto-Rheological dampers. The proposed approach concerns road adaptation using on-line road profile identification based on a non linear algebraic observer with unknown input. Then, the suspensions forces distribution in each corner of vehicle is performed considering roll dynamics. In this LPV/$\mathcal{H}_\infty$ strategy, 2 varying parameters are used to model the semi-active behaviour of the MR dampers, and 2 other ones, namely, the road roughness identification and roll dynamics, are considered for the road adaptation and the full vehicle vertical dynamics control.**

**Different ISO road classes are used to test the efficiency of the on-line non linear algebraic road profile identification.**

**Simulations scenarios, applied on a non linear full vehicle model, are used to evaluate the LPV/$\mathcal{H}_\infty$ controller performances in term of passengers comfort and road holding improvement in different driving situations.**

***Keywords*: algebraic identification methods, road profile estimation, algebraic observers with unknown input, LPV full vehicle conrol .**

## I. INTRODUCTION

The continuously expending vehicle market has led the automotive industry to develop more intelligent system to improve vehicle safety and passengers comfort. Every year, car's accident death rate is increasing (2.2% of the global mortality in 2009 by the *World Health Organisation*) due to loss of stability or manoeuvrability at high speed under critical road conditions.

Academic community has, also, been interested by developing new control strategies that enhances cars performances. In [1], a gain scheduling road adaptation control strategy is presented, however, the control synthesis is not oriented to reach on-line control objective adaptation since the road identification system is supposed available. Also, The existing estimation method for the road identification are either empirical rules [2] or extremely expensive since it needs a highly equipped vehicle [3]. In [4], the road roughness is estimated at a variable velocity by using different standardized roads (ISO 8608), but the *ANN-NARX* estimator could demand many computational resources for an online estimation.

Recently, authors have developed some novel control strategies for semi-active MR dampers. In [5], an LPV modeling and control of the semi-active MR dampers for automotive systems is introduced. Also, in [6], the design and analysis of an LPV semi-active suspension controller are presented.

In this paper, a new non linear estimation approach is used to identify the road profile. This estimation is used to adapt the control of the semi-active dampers to the road profile. Since the vehicle is considered to be equipped with the same MR dampers in the four corners, a roll adaptation suspension forces is provided to tune these dampers based on the load transfer that each corner supports. Finally, in this LPV/$\mathcal{H}_\infty$ strategy, four varying parameters are considered, 2 parameters to model the semi-activeness of the dampers, the other ones to tune the suspension actuators to the road profile and the supported loads.

This paper is organized as follows: Section 2 introduces the road profile estimation based on a non linear algebraic observer with unmown input. The section 3 tackles the LPV/$\mathcal{H}_\infty$ control with the road adaptation and the suspension semi-active forces distribution based on the roll dynamics. In section 4 simulations performed on the non linear full vehicle model equipped with four Magneto-Rheological dampers are compared with the passive case to emphasize the improvement brought by the proposed strategy. Finally, conclusions are given in the last section.

## II. ROAD PROFILE ESTIMATION METHOD

This section is devoted to the estimation of road profile used in the LPV controller for the semi-active suspension system of a vehicle. The estimation method uses the algebraic framework to design an algebraic observers with unknown inputs. This estimation method uses also a quarter of vehicle model of the suspension.

### A. Quarter vehicle model of suspension system

For the road profile estimation, a quarter of vehicle (QoV) model of a suspension system illustrated in Fig. 1 is used. This model describes the motions of the sprung and unsprung mass. The QoV system dynamics is governed by the following equation:

$$\begin{cases} m_s\ddot{z}_s = -k_s z_s + k_s z_{us} - \omega_1 \\ m_{us}\ddot{z}_{us} = -k_s z_s - (k_s + k_t) z_{us} + \omega_1 + k_t\omega_2 \end{cases} \quad (1)$$

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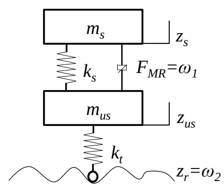



Fig. 1. Quarter vehicle model for a semi-active suspension

For (1), the damping force ($F_{MR} = \omega_1$) and the road profile ($z_r = \omega_2$) are assumed the unknown inputs for the algebraic observer design procedure.

### *B. Road profile estimation method based on algebraic observer with unknown input*

The estimation of the road profile is addressed in this section. The estimation method uses the algebraic framework devoted for the design of algebraic observers with unknown inputs [7], [8]. The estimation approach uses also the algebraic identification methods for the numerical differentiation of noisy signals [8], [9]. The estimation with unknown input is based on the following properties [7], [8]:

*Property 1:* the algebraic observability of any nonlinear system with unknown inputs is equivalent to express the dynamical state and the unknown inputs as functions of the inputs, the measured outputs and their finite time derivatives.

*Property 2:* A system is said observable with unknown inputs if, any state variable or an input variable, can be formulated as a function of the output and their finite time derivatives. This function can be called as an input-free estimator. It means in other words that an input-output system is observable with unknown input if, and only if, its zero dynamics is trivial. In addition, if the system is square, then the system is called flat[1] system with its flat output.

To establish the estimation method of road profile, we choose the displacements of sprung mass $z_s$ and unsprung mass $z_{us}$ as flat outputs.

$$y = \begin{bmatrix} y_1 \\ y_2 \end{bmatrix} = \begin{bmatrix} z_s \\ z_{us} \end{bmatrix} \tag{2}$$

The following algebraic observer estimation method of the road profile is established using the observability properties 1 and 2, the measured outputs (2) and the quarter vehicle model of suspension system (1):

$$\begin{cases} \hat{\dot{z}}_s = \dot{y}_1 \\ \hat{\dot{z}}_{us} = \dot{y}_2 \\ \hat{\omega}_1 = k_s y_2 - k_s y_1 - m_s \ddot{y}_1 \\ \hat{\omega}_2 = \frac{1}{k_t}\left(m_s \ddot{y}_1 + m_{us}\ddot{y}_2 + 2k_s y_1 + k_t y_2\right) \end{cases} \tag{3}$$

According to the properties 1 and 2, the system (1) is flat and the chosen outputs $y_1$ and $y_2$ are flat outputs.

[1]The differential flatness property of nonlinear systems in a differential algebraic context was introduced by [10], [11]

*Remark 2.1:* The algebraic observer with unknown inputs (3) is established thanks to the algebraic numerical differentiator (4) used to estimate the time derivatives of the measured flat outputs $y_1$ and $y_2$.

### *C. Algebraic identification*

To have an interesting filtering and numerical differentiation of noisy signals, the algebraic estimation techniques are used. This estimation is performed using the recent advances in [8], [9], which yield efficient real-time filters. For our study, these estimators are used to compute the derivatives of the flat outputs to design an input-free estimator of road profile. The following formulae may be used to estimate the $1^{\text{st}}$ order derivative of a signal $y(t)$:

$$\hat{\dot{y}}(t) = -\frac{3!}{h^3}\int_{t-h}^{t}(2h(t-\tau)-h)y(\tau)d\tau \tag{4}$$

Note that the sliding time window $[t-h,\, t]$ may be quite short and $h$ is sample time.

### *D. Diagram block of road profile estimation method*

The block diagram in Fig. 2 shows two parts of the road profile estimation method: the first one presents the filtering and numerical differentiation of the measured outputs, while the second one illustrates the road profile estimation using an algebraic observer with unknown input.

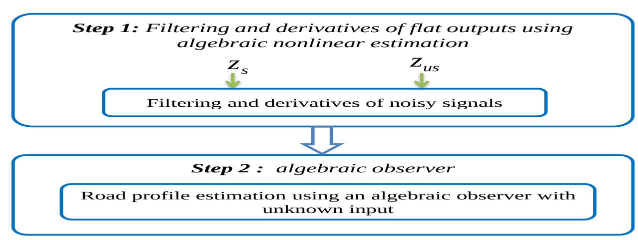



Fig. 2. Block diagram of the road profile estimation method

## III. ALGEBRAIC OBSERVER ESTIMATION RESULTS

To test the efficiency of proposed algebraic observer with unknown input, a quarter-car model is considered subject to a measured road profile excitation (see Fig. 4). The measured output signals considered as the flat output of the algebraic observer are $z_s$ and $z_{us}$ (the chassis displacement and wheel motion, resp), as shown in Fig. 3.

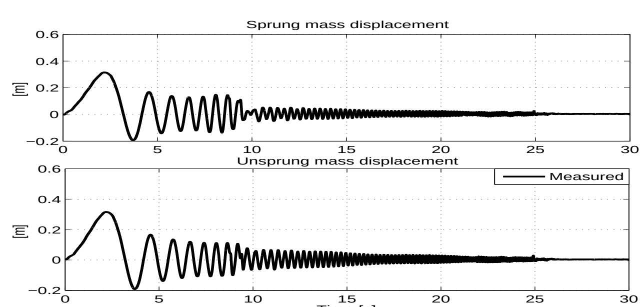



Fig. 3. Used flat outputs: sprung and unsprung mass displacements

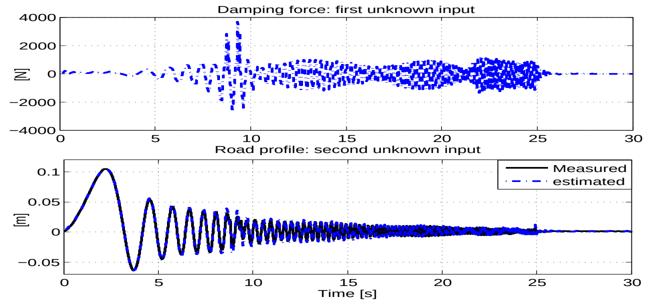



Fig. 4. Unknown inputs estimation: damping force and road profile ($\hat{\omega}_1$, $\hat{\omega}_1$, resp)

Fig. 4 shows the efficiency of the proposed non linear algebraic estimation. The observer estimates perfectly the considered road profile. It will be used in the control strategy, designed below.

## IV. LPV/$\mathcal{H}_\infty$ Semi-Active Suspension Controller Synthesis

In this work, since the vehicle is equipped with the same MR dampers on the four corners of the car, the LPV/$\mathcal{H}_\infty$ Semi-Active Suspension Controller is designed using the quarter vehicle LPV model, as in [5]. The distribution of the suspension forces is performed using the evaluation of the load transfer, given by the roll dynamics, as a varying parameter in the road adaptation control strategy. This strategy is also based on the road profile estimation given by the non linear algebraic observer with unknown input.

### A. LPV model formulation

In this paper, a quarter-car LPV model is considered following [5], [12]. It includes a MR damper model of the form:

$$F_{MR} = I f_c \tanh\left(a_1 \dot{z}_{def} + a_2 z_{def}\right) + b_1 \dot{z}_{def} + b_2 z_{def} \quad (5)$$

where $I$ is the input current and $a_1$, $a_2$, $b_1$ and $b_2$ are some real parameters characterizing the MR damper.
As shown in [5], [12], a quarter-car model (1) including a MR damper model (5) can be represented in the LPV form with 2 varying parameters:

$$\begin{cases} \dot{x}_{lpv} = A_{lpv}\left(\rho_1, \rho_2\right) x_{lpv} + B_1 u_c + B_2 w_2 \\ y_{lpv} = C_1 x_{lpv} \end{cases} \quad (6)$$

where

$$x_{lpv} = \begin{pmatrix} x_s \\ x_f \end{pmatrix}^T,$$
$$A_{lpv}\left(\rho_1, \rho_2\right) = \begin{pmatrix} A_s + \rho_2 B_{s2} C_{s2} & \rho_1 B_s C_f \\ 0 & A_f \end{pmatrix},$$
$$B_1 = \begin{pmatrix} 0 \\ B_f \end{pmatrix}, B_2 = \begin{pmatrix} B_{s1} \\ 0 \end{pmatrix}, C_1 = \begin{pmatrix} C_s \\ 0 \end{pmatrix}^T$$
$$\rho_1 = \tanh(C_{s2} x_s) \tanh\left(\frac{C_f x_f}{F_1}\right) \frac{F_1}{C_f x_f},\ \rho_2 = \frac{\tanh(C_{s2} x_s)}{C_{s2} x_s}$$

$x_s$, $A_s$, $B_s$, $B_{s1}$, $B_{s2}$, $C_s$ and $C_{s2}$ are state and matrices of a state-space representation of the $QoV$ model; $x_f$, $A_f$, $B_f$, $C_f$ are state and matrices of a representation of the low-pass filter $W_{filter} = \omega_f/(s + \omega_f)$, which is added to the plant to make the control input matrices parameter independent, considering $z_{def}$ and $\dot{z}_{def}$ as output.

### B. LPV/$\mathcal{H}_\infty$ control synthesis

The main contribution of this paper is the use of a 2 additional varying parameters in the control synthesis: the first one ($\rho_3$), that schedules the suspension actuator work according to a new algebraic road estimation strategy, as shown in Fig. 5. A LPV controller is proposed following the $\mathcal{H}_\infty$ control configuration in Fig. 5, where the weighting functions are the one given in [5]:

- $W_{filter} = \omega_f/(s + \omega_f)$, with a large bandwidth to decouple the input and the varying parameters.
- $W_{\ddot{z}_s} = k_{\ddot{z}_s} \frac{s^2 + 2\zeta_{11}\omega_{11}s + \omega_{11}^2}{s^2 + 2\zeta_{12}\omega_{12}s + \omega_{12}^2}$, to account for passengers comfort at low frequencies.
- $W_{z_{us}} = k_{z_{us}} \frac{s^2 + 2\zeta_{21}\omega_{21}s + \omega_{21}^2}{s^2 + 2\zeta_{22}\omega_{22}s + \omega_{22}^2}$, to account for road holding at high frequencies.
- $W_{z_r} = 5 \times \rho_3 \times 10^{-2}$, road profile gain also scheduled by the road roughness estimation to adapt the control synthesis.

In addition to the 2 parameters $\rho_1$, $\rho_2$ representing the damper non linearities, 2 new parameters are indeed introduced. The first one ($\rho_3$) schedules the suspension actuator according to the new algebraic road estimation strategy. Indeed, depending on the value of the road roughness, the suspension control is adapted to meet the required performances.

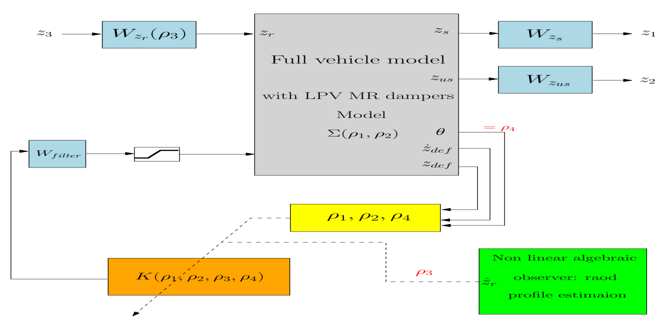



Fig. 5. Suspension control design scheme.

Thus, the corresponding generalized plant is a 3 linear parameter depending system as follows:

$$\Sigma_{gv}(\rho_1, \rho_2, \rho_3) := \begin{cases} \dot{\xi} = A(\rho_1, \rho_2, \rho_3)\xi + B_1 u + B_2 \tilde{w} \\ \tilde{z} = \tilde{C}_1 \xi + D_{11} u + D_{12} \tilde{w} \\ y = \tilde{C}_2 \xi + D_{21} u + D_{22} \tilde{w} \end{cases} \quad (7)$$

where $\xi = [\chi_{vert}\ \ \chi_w]^T$; $\tilde{z} = [z_1\ \ z_2\ \ z_3]^T$; $\tilde{w} = z_r$; $y = [z_{def}\ \ \dot{z}_{def}]^T$; $u = u_{ij}^{\mathcal{H}_\infty}$; $\chi_{vert}$ are the states in the vertical dynamics of the $QoV$ model and $\chi_w$ are the vertical weighting functions states.

The second one represents an interesting contributions is the use of the roll dynamics as a varying parameter $\rho_4 = \theta$ to schedule the distribution of the left & right suspensions on the four corners of the vehicle and tune the suspension dampers smoothly, thanks to the LPV frame work, from "soft" to "hard" to improve the car performances according to the driving situation. This distribution is handled using a specific structure of the suspension controller, given as follows :

$$K_s(\rho_n) := \begin{cases} \dot{x}_c(t) = A_c(\rho_n)x_c(t) + B_c(\rho_n)y(t) \\ \begin{pmatrix} u_{fl}^{\mathcal{H}_\infty}(t) \\ u_{fr}^{\mathcal{H}_\infty}(t) \\ u_{rl}^{\mathcal{H}_\infty}(t) \\ u_{rr}^{\mathcal{H}_\infty}(t) \end{pmatrix} = \underbrace{U(\rho_4)C_c^0(\rho_4)}_{C_c(\rho_4)} x_c(t) \end{cases} \tag{8}$$

where

$$U(\rho_4) = \begin{pmatrix} 1-|\rho_4| & 0 & 0 & 0 \\ 0 & |\rho_4| & 0 & 0 \\ 0 & 0 & 1-|\rho_4| & 0 \\ 0 & 0 & 0 & |\rho_4| \end{pmatrix} \tag{9}$$

$\rho_n$, $n = 1, 2, 3$ (more explanation on the varying parameter used in the synthesis is given in the following), $x_c(t)$ is the controller state, $A_c(\rho_n)$, $B_c(\rho_n)$ and $C_c(\rho_4)$ are the state matrices of the controller. $u^{\mathcal{H}_\infty}(t) = [u_{fl}^{\mathcal{H}_\infty}(t)u_{fr}^{\mathcal{H}_\infty}(t)u_{rl}^{\mathcal{H}_\infty}(t)u_{rr}^{\mathcal{H}_\infty}(t)]$ the input control of the suspension actuators and $y(t) = z_{def}(t)$.

*Remark 1:* In this synthesis, the authors stress that one interesting innovative point in this approach is the use of a fixed structure controller, but a parameter dependency on the control output matrix is introduced to allow the accurate suspension force on every corner of the vehicle , depending on the driving situation, to achieve the performance objectives.

### C. Scheduling parameters

The control of the vertical dynamics is ensured by the suspension system in order to achieve frequency specification performances, [13]. In this study, both the considered *QoV* model for synthesis and the controller are parameters depending; $\rho_1$ and $\rho_2$ present in the vehicle model ensures the semi-active characteristic of the MR dampers and the saturation limitations.

The varying parameter $\rho_3$ allows an on-line adaptation of the semi-active suspension system to road profiles, and is given by:

$$\rho_3 = K_{\rho_3} \cdot S_{z_r}(f_i) \quad \in \quad [0, 1] \tag{10}$$

where

- $\cdot S_{z_r}(f_i)$ is the road roughness reconstructed based on the non linear algebraic road profile estimation, using Fourier series to estimate the amplitude and the frequency of the road profile, for more details see [14].
- $K_{\rho_3}$ is used to bound $\rho_3$, such that

$$I(\rho_3) := \begin{cases} I = I_{max} & if \quad \rho_3 \to \overline{\rho_3} \\ I_{min} < I < I_{max} & if \quad \underline{\rho_3} < \rho_3 < \overline{\rho_3} \\ I = I_{min} & if \quad \rho_3 \to \underline{\rho_3} \end{cases} \tag{11}$$

This road adaptive control strategy allows to provide a good on-line trade-off between road holding and passengers comfort, which are conflicting objectives, such that:

- When $\rho_3$ is high, the road roughness is high, and the semi-active damper is tuned to be harder ($u^{\mathcal{H}_\infty} \to I_{max}$) in order to improve the car road holding and guarantee the vehicle safety at high velocities or comfort at low velocities.
- Conversely, when $\rho_3$ is low, the road roughness is low, and the *MR* damper is set to be softer to enhance comfort at low velocities or road holding at high velocities.

The varying parameter $\rho_4$ ensures the accurate distribution of the suspension forces, based on the load distribution of the vehicle. Here, left & right load transfer is considered. To ensure this distribution, the roll dynamics are used to schedule the semi-active suspensions effort ($\rho_4 = \theta$). This will alow to optimise the use of the different suspensions actuators to enhance vertical car's dynamics.

*Remark 2:* The proposed *LPV/$\mathcal{H}_\infty$* robust controller is synthesized by using *LMIs* solution for polytopic systems; all varying parameters are considered bounded: $\rho_1 \in [-1, 1]$, $\rho_2 \in [0, 1]$, $\rho_3 \in [0, 1]$ and $\rho_4 \in [-1, 1]$.

## V. Simulation Results

Fig. 6 represents the implementation scheme for the proposed LPV/$\mathcal{H}_\infty$ strategy based on algebraic estimation:

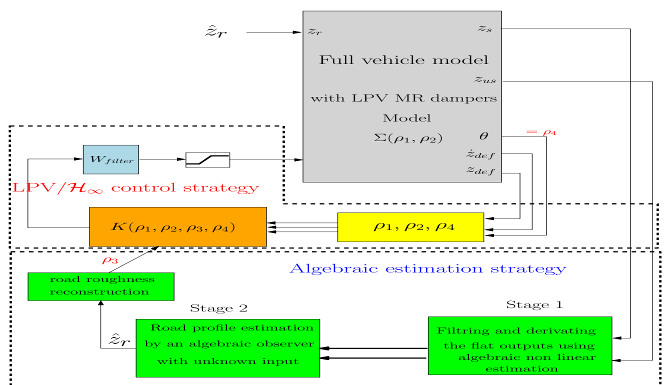



Fig. 6. Implementation scheme of the proposed LPV/$\mathcal{H}_\infty$.

Simulations are performed on a non linear full vehicle non linear model [15]. The model parameters are those of a Renault Mégane Coupé, obtained during a collaborative study with the MIPS laboratory in Mulhouse, through identification with real data. The full vehicle model is used to emphasize the effect of the varying parameter $\rho_4 = \theta$ (roll dynamics) for the distribution of the suspensions forces to enhance passengers comfort and the road holding of the vehicle. Then, the road profile estimated in Fig. 4 is applied in the left side of the vehicle, namely, under the front left wheel $z_{us_{fr}}$ and the rear left wheel $z_{us_{rr}}$. To prove the efficiency of the estimation strategy, the following ISO road are used to test it: The following results are obtained:

TABLE I

Classification of road profiles (ISO 8608).

| Type of Road | Class | Lower $c_r$ $m^2(cycles/m)$ | Upper $c_r$ $m^2(cycles/m)$ |
|---|---|---|---|
| Smooth runway | A | $3.2 \times 10^{-7}$ | $1.6 \times 10^{-14}$ |
| Smooth highway | B | $3.2 \times 10^{-7}$ | $1.2 \times 10^{-6}$ |
| Highway with gravel | C | $1.2 \times 10^{-6}$ | $5.1 \times 10^{-6}$ |
| Rough runway | D | $5.1 \times 10^{-6}$ | $2.0 \times 10^{-5}$ |
| Pasture | E | $2.0 \times 10^{-5}$ | $8.2 \times 10^{-5}$ |
| Plowed field | F | $8.2 \times 10^{-5}$ | $3.3 \times 10^{-4}$ |

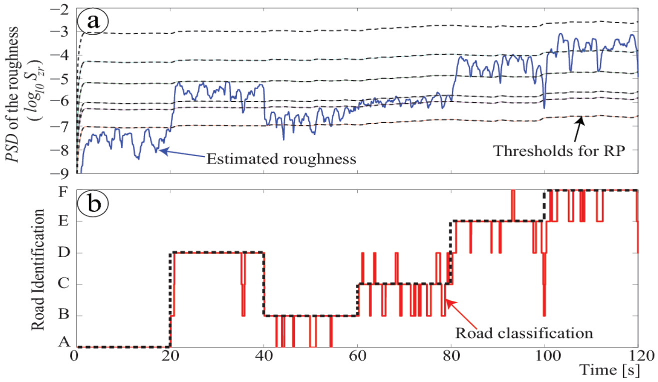
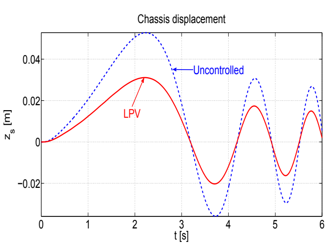



Fig. 7. On-line roughness estimation (up) and final result in the road identification algorithm (bottom).

In Fig. 7, it can be noticed that the road profile identification is well reached by the non linear algebraic observer. To show the efficiency of the proposed LPV/$\mathcal{H}_\infty$ full vehicle semi-active suspension control strategy, two scenarios are proposed.

### A. *First simulation scenario*

The first one concerns the vehicle running on a plowed field (see Table.V) at $v_x = 30$ kmh. In these case the predominating performance objective is the passengers comfort.



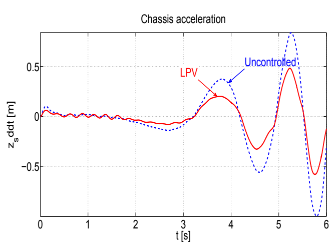
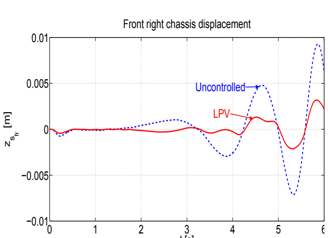




Fig. 8. Chassis displacement of the gravity center $z_s$.

Fig. 9. Chassis acceleration of the gravity center $\ddot{z}_s$.

Fig. 8 and Fig. 9 show the chassis displacement in center of gravity (chassis acceleration in the center of gravity respectively). It can be noticed that the proposed LPV/$\mathcal{H}_\infty$ controller (in red) enhance better these dynamics representative of the passenger comfort than in the case of the passive suspension system (in blue). Actually, by calculating the *RMS* (root mean square) of the chassis displacement ($z_s$) and the chassis acceleration ($\ddot{z}_s$), one can notice an improvement of 30% using the LPV/$\mathcal{H}_\infty$ control of the MR dampers.



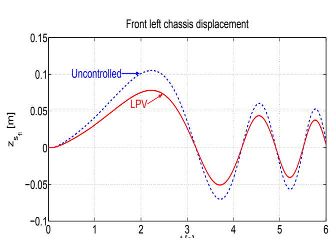
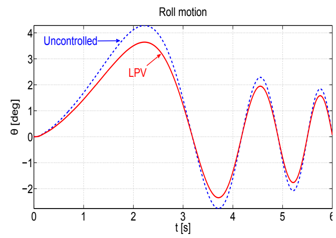




Fig. 10. Front right chassis displacement $z_{s_{fr}}$.

Fig. 11. Front left chassis displacement $z_{s_{fl}}$.

In Fig. 10 and Fig. 11, the front right and left chassis displacement is shown. The effect of the suspensions forces distribution can be seen. Since the road profile is applied on the left side of the vehicle, it is clear that a larger load is applied in the right side, and then with the LPV scheduling strategy larger suspensions forces can be applied on this side. Here, one can see that chassis displacement is better attenuated on the right corners of the vehicle than on left ones which copes with the proposed approach objectives.



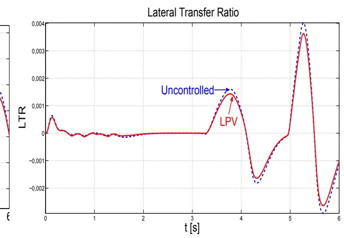
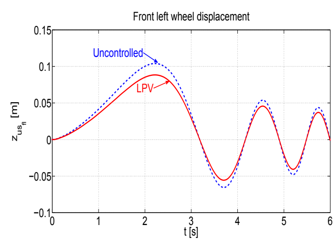




Fig. 12. Roll motion $\theta$.

Fig. 13. Load transfer ratio, LTR.

Fig. 12, Fig. 13, Fig. 14 and Fig. 15 are roll motion, lateral transfer ratio, front left wheel and front right wheel displacement respectively, representative of the road holding of the vehicle. The improvement brought on these dynamics can be noticed, even if less than for the passengers comfort car's dynamic.



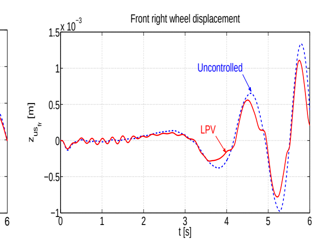



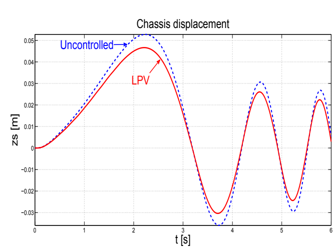

Fig. 14. Front left wheel displacement $z_{us_{fl}}$.

Fig. 15. Front right wheel displacement $z_{us_{fr}}$.

Indeed, calculating the *RMS* of the roll motion and the right wheel displacement proves that they have been improved up of 15% and the lateral transfer ratio and the left wheel displacement up to 5% compared to the passive case.

### B. *Second simulation scenario*

In the second scenario, the vehicle runs at a high speed, $v_x = 100$ kmh, on a smooth road (ISO road A, see Table.V). Under these conditions, the predominating performance objective is the vehicle road holding ($\theta$ and $z_{us}$).



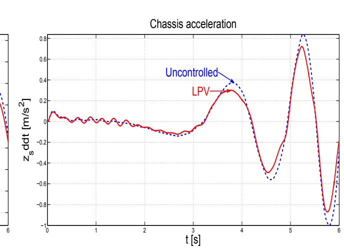




Fig. 16. Chassis displacement of the gravity center $z_s$.

Fig. 17. Chassis acceleration of the gravity center $\ddot{z}_s$.

Fig. 16 and Fig. 17 show the chassis displacement in center of gravity (chassis acceleration in the center of gravity respectively). Calculating the *RMS* of these signals resulting on the LPV/$\mathcal{H}_\infty$ controller shows an improvement of $10\%$ compared to the passive case. The improvement seems small since the controller, under these driving conditions, is oriented to enhance the road holding of the vehicle.

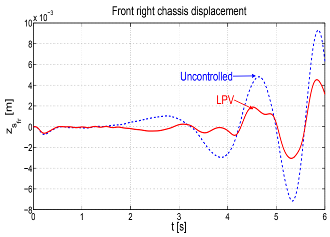



Fig. 18. Front right chassis displacement $z_{s_{fr}}$.

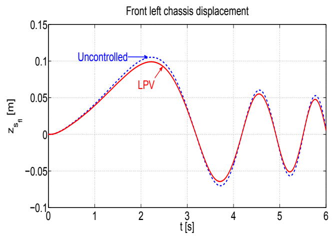



Fig. 19. Front left chassis displacement $z_{s_{fl}}$.

Fig. 18 and Fig. 19 show the front right and left chassis displacement. It can be noticed that the right chassis displacement is better attenuated than the left one, even if the improvement remains small. This can be explained by the semi-active suspension forces distribution process thanks the varying parameter $\rho_4 = \theta$.

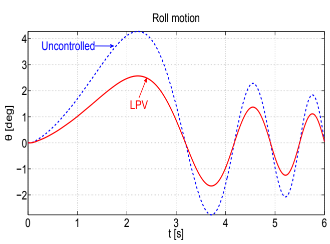



Fig. 20. Roll motion $\theta$.

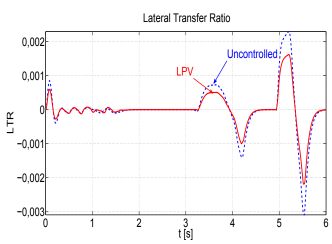



Fig. 21. Load transfer ratio, LTR.

Fig. 20, Fig. 21, Fig. 22 and Fig. 23 are roll motion, lateral transfer ratio, front left wheel and front right wheel displacement respectively. Here, by calculating the *RMS* of each signal, it can be seen that the roll motion is enhanced up to "$40\%$" compared to the passive case. Also the lateral transfer ratio is decreasing by "$30\%$".

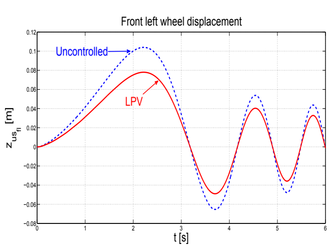



Fig. 22. Front left wheel displacement $z_{us_{fl}}$.

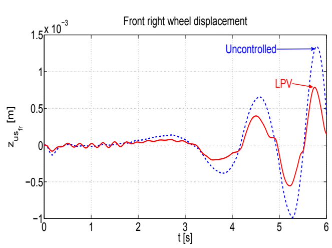



Fig. 23. Front right wheel displacement $z_{us_{fr}}$

The front right and left wheel displacement is improved up to "$40\%$ (resp. up to "$25\%$) compared to the uncontrolled vehicle.This improvement, which concerns the car road holding induces better stability and safety for the vehicle in critical driving situations.

## VI. Conclusions

In this work, a new LPV/$\mathcal{H}_\infty$ full vehicle control strategy for semi-active suspension systems has been presented. The main objective is to adapt the vertical behaviour of the vehicle to the road profile and to tune each one of the four semi-active Maneto-Rhelogical dampers based on the load transfer. A non linear algebraic observer with unknown input is used to estimate the road profile and provides the varying parameter for the road control adaptation. Also, the suspension forces distribution is ensured by scheduling the outputs of the semi-active suspension controller from the roll dynamics $\theta$ (representing the right & left load transfer) when performing different driving scenarios.

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